

Notice No.9

Rules and Regulations for the Classification of Ships, July 2017

The status of this Rule set is amended as shown and is now to be read in conjunction with this and prior Notices. Any corrigenda included in the Notice are effective immediately.

Please note that corrigenda amends to paragraphs, Tables and Figures are not shown in their entirety.

Issue date: June 2018

Amendments to	Effective date	IACS/IMO implementation (if applicable)
Part 5, Chapter 1, Sections 3 & 5	1 July 2018	1 July 2018
Part 5, Chapter 2, Sections 3 & 11	1 July 2018	1 July 2018
Part 5, Chapter 2, Sections Scope, 1, 4, 7, 8 & 9	1 July 2018	N/A
Part 5, Chapter 24, Section 9	1 July 2018	N/A
Part 6, Chapter 2, Sections 1, 9, 11 & 19	1 July 2018	N/A
Part 7, Chapter 11, Sections 1, 2 & 3	1 July 2018	N/A
Part 7, Chapter 13, Sections 3 & 6	1 July 2018	N/A



Lloyd's
Register

Working together
for a safer world

Part 5, Chapter 1

General Requirements for the Design and Construction of Machinery

■ **Section 3**

Operating conditions

3.9 Astern power

3.9.1 Sufficient astern power is to be provided to maintain control of the ship in all normal circumstances.

3.9.2 Astern turbines are to be capable of maintaining in free route astern 70 per cent of the ahead revolutions, corresponding to the maximum propulsion shaft power for which the machinery is to be classed, for a period of at least 30 minutes without undue heating of the ahead turbines and condensers.

3.9.1 In order to maintain sufficient manoeuvrability and secure control of the ship in all normal circumstances, the main propulsion machinery is to be capable of reversing the direction of thrust so as to bring the ship to rest from the maximum service speed. The main propulsion machinery is to be capable of maintaining in free route astern at least 70 per cent of the ahead revolutions corresponding to the maximum continuous ahead power for which the vessel is classed.

3.9.2 Where steam turbines are used for main propulsion, they are to be capable of maintaining in free route astern at least 70 per cent of the ahead revolutions corresponding to the maximum continuous ahead power for which the vessel is classed for a period of at least 15 minutes.

■ **Section 5**

Trials

5.2 Sea trials

(Part only shown)

5.2.2 The trials are to include demonstrations of the following:

(c) The ability to permit astern running at 70 per cent of the full power ahead revolutions corresponding to the maximum continuous ahead power for which the vessel is classed.
(e)(d) In steam turbine installations, the ability to permit astern running at 70 per cent of the full power ahead revolutions corresponding to the maximum continuous ahead power for which the vessel is classed without adverse effects. This astern trial need only be of 15 minutes' duration, but may be extended to 30 minutes at the Surveyor's discretion. To avoid overheating of the turbine due to the effects of 'windage' and friction, the astern trial is not to exceed 30 minutes or the manufacturer's recommendation.

5.2.3 Main propulsion systems are to undergo tests to demonstrate the astern response characteristics. The tests are to be carried out over at least the manoeuvring range of the propulsion system and from all control positions. A test plan is to be provided by the yard and accepted by the Surveyor. If specific operational characteristics have been defined by the manufacturer, then these are to be included in the test plan.

5.2.4 The reversing characteristics of the propulsion plant, including the blade pitch control system of controllable pitch propellers, are to be demonstrated and recorded during trials.

Existing paragraphs 5.2.3 to 5.2.7 have been renumbered 5.2.5 to 5.2.9.

5.2.10 For main propulsion systems with reversing gears, controllable pitch propellers or electric propeller drive, running astern is not to lead to the overload of propulsion machinery.

Existing paragraph 5.2.8 has been renumbered 5.2.11.

Part 5, Chapter 2

Reciprocating Internal Combustion Engines

Scope

Scope Section has been deleted and the contents moved into sub-Sections 1.1 and 1.2 of Section 1 as shown below.

■ Section 1

General requirements

1.1 Application

1.1.1 Engines providing power for services essential to the safety of the vessel are to be constructed under survey and in accordance with the requirements of this Chapter (see also Pt 1, Ch 2, 2.4 Class notations (machinery) 2.4.1).

1.1.2 The requirements of this Chapter are applicable to reciprocating internal combustion engines operating on liquid, gas or dual fuel for main propulsion and essential auxiliary services (hereinafter referred to as engines). Pt 5, Ch 2, 3 Crankshaft Design is not applicable to auxiliary engines having powers of less than 110 kW.

1.2 Scope

1.2.1 For the purposes of this Chapter engine type, expressed by the manufacturer/licensor's designation, is defined by:

- (a) the bore and stroke;
- (b) the method of injection (i.e. direct injection, indirect injection, pilot injection);
- (c) the fuel pump and injection system (independent line to fuel oil valve, common rail);
- (d) the valve and injection operation (by cams or electronically controlled);
- (e) the fuel(s) used (liquid, dual-fuel, gaseous, etc.);
- (f) the working cycle (4-stroke, 2-stroke);
- (g) the gas exchange (naturally aspirated, turbocharged, etc.);
- (h) the method of turbocharging (pulsating system, constant pressure system);
- (i) the charging air cooling system (with or without intercooler, number of stages);
- (j) cylinder arrangement (in-line, vee, etc.);
- (k) the maximum continuous power per cylinder (or maximum continuous brake mean effective pressure) at maximum continuous speed;
- (l) the manufacturer and type of governor (and control system if applicable) fitted.

1.2.2 A complete engine includes the control system, turbocharger(s) and all ancillary systems and equipment referred to in this Chapter that are used for operation of the engine for which there are rule requirements; this includes systems allowing the use of different fuel types.

1.2.3 Arrangements for dual fuel engines will be specially considered.

1.2.4 Primary exhaust gas emissions abatement plant (where fitted) is to meet the requirements of this Chapter; additionally, it is to meet the requirements of Pt 5, Ch 24 Emissions Abatement Plant for Combustion Machinery. Where secondary exhaust gas emissions abatement systems are fitted to engines, they are to meet the requirements of Pt 5, Ch 24 Emissions Abatement Plant for Combustion Machinery.

Existing sub-Sections 1.1 and 1.2 have been renumbered 1.3 and 1.4.

1.1.3 Approval process

1.1.3.2 Each complete engine, as defined in the scope Pt 5, Ch 2, 1.2 Scope, intended for installation on an LR Classed vessel, is to have an LR Engine Certificate.

(Part only shown)

1.1.3.4 To apply for an LR Engine Certificate, the following are to be submitted:

- (a) a list of all documents identified in the 'for information' and 'for appraisal' columns of Table 2.1.1 Plans and particulars to be submitted with the relevant drawing numbers and revision status. This list is to cross-reference the approved plans previously submitted in accordance with Pt 5, Ch 2, 1.1 Approval process 1.1.4.(a) as part of the engine Type Approval and identify any plans that have been modified.
- (b) where there is a licensor/licensee arrangement the list required by 1.1.4.(a) Pt 5, Ch 2, 1.1 Approval process 1.3.4(a) is to cross-reference the drawings submitted by the designer in accordance with Pt 5, Ch 2, 1.1 Approval process 1.1.4.(a) as part of the engine Type Approval. This list is to identify all changes where the approved design has been modified by the licensee. Where the licensee proposes design modifications to components, a statement is to be made confirming the licensor's acceptance of the proposed changes. If designer/licensor's acceptance is not confirmed, the engine is to be regarded as a different type and is subject to the complete appraisal and type approval process.

4.1.6 1.3.6 For appraisal of emergency generators engines and turbochargers additional submissions are required. See Pt 5, Ch 2, 4.2 1.4 *Submission requirements* 1.2.4 1.4.4 and Pt 5, Ch 2, 1.2-1.4 *Submission requirements* 1.2.5-1.4.5 as applicable.

4.2 1.4 Submission requirements

Existing paragraph 1.2.1 has been renumbered 1.4.1.

(Part only shown)

4.2-2 1.4.2 A schedule of testing at engine builders' packager's or system integrator's facility, pre-sea trial commissioning and sea trials is to be submitted. The test schedules are to identify all modes of engine operation and the sea trials are to include typical port manoeuvres under the intended engine operating modes. The schedule is to include:

Paragraphs 1.2.3 to 1.2.8 have been renumbered 1.4.3 to 1.4.8.

■ Section 3 Crankshaft Design

Existing Section 3 has been deleted in its entirety and replaced with below.

3.1 Application

3.1.1 Pt 5, Ch 2, 3 Crankshaft Design is not applicable to auxiliary engines having powers of less than 110 kW.

3.2 Scope

3.2.1 The formulae given in this Section are applicable to solid or semi-built crankshafts of forged or cast steel, having a main support bearing adjacent to each crankpin.

3.2.2 This section uses the static determinate method; alternative methods, including a fully documented stress analysis, will be specially considered.

3.2.3 Calculations are to be carried out for the maximum continuous power rating for all intended operating conditions.

3.2.4 Designs of crankshafts not included in this scope will be subject to special consideration.

3.2.5 Where a crankshaft design involves the use of surface treated fillets, or when fatigue parameter influences are tested, or when working stresses are measured, the relevant documents with calculations/analysis are to be submitted to LR.

3.2.6 The design of crankshafts is based on an evaluation of safety against fatigue in the highly stressed areas. The calculation is also based on the assumption that the areas exposed to highest stresses are:

- fillet transitions between the crankpin and web as well as between the journal and web,
- outlets of crankpin oil bores.

3.2.7 When the journal diameter is equal to or larger than the crankpin one, the outlets of main journal oil bores are to be formed in a similar way to the crankpin oil bores, otherwise separate documentation of fatigue safety will be specially considered.

3.2.8 Calculation of crankshaft strength consists initially in determining the nominal alternating bending (see Pt 5, Ch 2, 3.6 *Calculation of bending stresses*) and nominal alternating torsional stresses (see Pt 5, Ch 2, 3.7 *Calculation of torsional stresses*) which, multiplied by the appropriate stress concentration factors (SCF) (see Pt 5, Ch 2, 3.8 *Stress concentration factors*), result in an equivalent alternating stress (uniaxial stress) (see Pt 5, Ch 2, 3.10 *Equivalent alternating stress*). This equivalent alternating stress is then compared with the fatigue strength of the selected crankshaft material (see Pt 5, Ch 2, 3.11 *Fatigue strength*). This comparison will show whether or not the crankshaft concerned is dimensioned adequately (see Pt 5, Ch 2, 3.12 *Acceptability criteria*).

3.2.9 Further information and guidance for crankshaft design is provided in the LR Guidance Notes *Guidance Notes for Crankshaft SCF Calculation using Finite Element Method* and *Guidance for the Evaluation of Crankshaft Fatigue Tests*.

3.3 Information to be submitted

3.3.1 For the calculation of crankshafts, the documents and particulars listed below are required, this information is provided by completing LR Form 2073 and submitting the applicable plans required in *Table 2.1.1 Plans and particulars to be submitted*:

- Crankshaft drawing (which must contain all data in respect of the geometrical configurations of the crankshaft);
- Type designation and kind of engine (in-line engine or V-type engine with adjacent connecting rods, forked connecting rod or articulated-type connecting rod);
- Operating and combustion method (2-stroke or 4-stroke cycle/direct injection, precombustion chamber, etc.) Number of cylinders;
- Output power at maximum continuous rating (MCR), in kW;

- Output speed at maximum continuous power, in rpm;
- Maximum firing pressure, P_{\max} , in MPa;
- Mean indicated pressure, in MPa;
- Charge air pressure (before inlet valves or scavenge ports, whichever applies), in MPa;
- Digitised gas pressure/crank angle cycle for MCR (presented at equidistant intervals at least every 5° CA);
- Mean piston speed;
- Compression ratio;
- Vee angle α_v , in degrees;
- Firing order numbered from driving end, see *Figure 2.3.1 Designation of cylinders*;
- Direction of rotation;
- Cylinder diameter, in mm;
- Piston stroke, in mm;
- Centre of gravity of connecting rod from large end centre, in mm;
- Radius of gyration of connecting rod, in mm;
- Length of connecting rod between bearing centres, L_H , in mm;
- Mass of single crankweb (indicate if webs either side of pin are of different mass values), in kg;
- Centre of gravity of crankweb mass from shaft axis, in mm;
- Mass of counterweights fitted (for complete crankshaft) indicate positions fitted, in kg;
- Centre of gravity of counterweights (for complete crankshaft) measured from shaft axis, in mm;
- All individual reciprocating masses acting on one crank, in kg;
- Crankshaft material specification(s) (according to ISO, EN, DIN, AISI, etc.);
- Mechanical properties of material (minimum values obtained from longitudinal test specimens):
 - tensile strength, in N/mm²
 - yield strength, in N/mm²
 - reduction in area at break, percentage
 - elongation, percentage
- Method of manufacture (free form forged, continuous grain flow forged, drop-forged, etc., with description of the forging process);
- For semi-built crankshafts – minimum and maximum diametral interference, in mm; and
- Particulars of alternating torsional stress calculations (see Pt 5, Ch 2, 3.7 Calculation of torsional stresses).

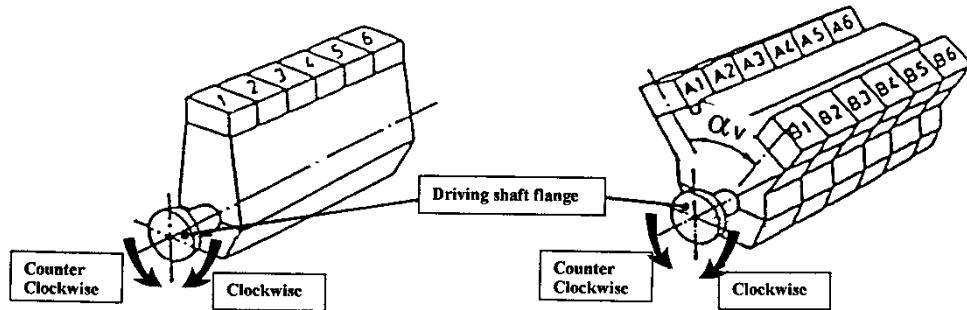


Figure 2.3.1 Designation of cylinders

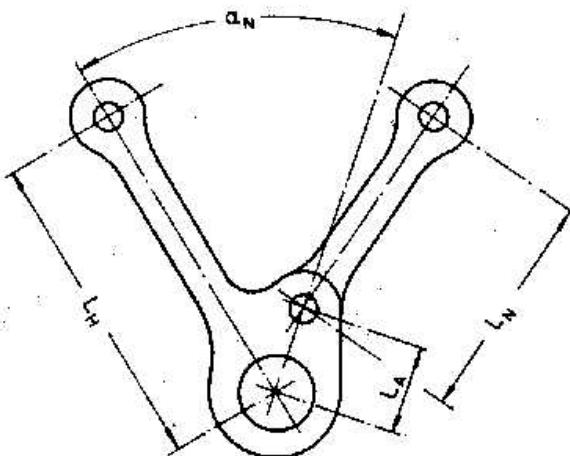


Figure 2.3.2 Articulated-type connecting rod

3.3.2 The following information is also required for appraisal of the crankshaft (not contained in Form 2073):

- For engines with articulated-type connecting rod (see *Figure 2.3.2 Articulated-type connecting rod*):
 - Distance to link point L_A , in mm
 - Link angle α_N , in degrees
 - Connecting rod length L_N , in mm
- firing interval (if applicable) i.e. if not evenly distributed
- Mass of connecting rod (including bearings), in kg;
- Mass of piston (including piston rod and crosshead where applicable), in kg;
- Every surface treatment affecting fillets or oil holes shall be specified so as to enable calculation according to Chapter 2 of the *LR Guidance Notes for Crankshaft SCF Calculation using Finite Element Method*;
 - This is to include Crankshaft fatigue enhancement factors K_1 and K_2 where applicable.
- Maximum alternating torsional stress τ_A (N/mm²)
- Mechanical properties of material (minimum values obtained from longitudinal test specimens), in addition to the information listed above:

Impact energy K_V , in Joules

3.4 Symbols

3.4.1 For the purposes of this Chapter the following symbols apply, see also *Figure 2.3.3 Crank dimensions for overlapped crankshaft*, *Figure 2.3.4 Crank dimensions for crankshaft without overlap*, *Figure 2.3.5 Crankpin section through the oil bore* and *Figure 2.3.6 Crankthrow of semi-built crankshaft*.

B = transverse breadth of web, in mm

D = crankpin diameter, in mm

D_A = the outside diameter of web or twice the minimum distance between centre-line of journals and outer contour of web, whichever is less, in mm

D_{BH} = diameter of axial bore in crankpin

D_{BG} = diameter of axial bore in journal

D_G = journal diameter

D_o = diameter of radial oil bore in crankpin, in mm

D_s = shrink diameter of main journal in web, in mm

E = pin eccentricity

E_m = Young's modulus of crankshaft material, in N/mm²

F = area related to cross-section of web, in mm²

K_e = bending stress factor (considers the influence of adjacent crank and bearing restraint)

K = fatigue enhancement factor ($K = K_1, K_2$)

K_1 = fatigue enhancement factor due to manufacturing process

K_2 = fatigue enhancement factor due to surface treatment

L_s = length of shrink fit, in mm

M_{BON} = alternating bending moment calculated at the outlet of crankpin oil bore

M_{BRFN} = alternating bending moment related to the centre of the web, in Nm

M_{TN} = maximum alternating torque, in Nm

M_{Tmax} = maximum value of the torque, in Nm

M_{Tmin} = minimum value of the torque, in Nm

Q_{RFN} = alternating radial force related to the web, in N

R_H, R_G = fillet radius at junction of web and pin or journal, in mm

S = pin overlap, in mm $S = \frac{D+D_G}{2} - E$

T_H, T_G = recess of pin or journal fillet radius into web measured from web face, in mm

W = axial thickness of web, in mm

W_{eqw} = section modulus related to cross-section of web, in mm^3

W_e = section modulus related to cross-section of axially bored crankpin, in mm^3

W_p = polar section modulus related to cross-section of axially bored crankpin or bored journal, in mm^3

y = distance between the adjacent generating lines of journal and pin, in mm

Note: $y \geq 0,05D_s$, where y is less than $0,1D_s$ special consideration is to be given to the effect of the stress due to the shrink fit on the fatigue strength at the crankpin fillet

α_B = bending stress concentration factor for crankpin fillet

α_T = torsional stress concentration factor for crankpin fillet

β_B = bending stress concentration factor for main journal fillet

Note. α_B and β_B are defined as the ratio of the maximum equivalent stress (von Mises) occurring in the fillets under bending load, to the nominal bending stress related to the web cross-section. See *Figure 2.3.7 Stress concentration factors in crankshaft fillets*.

β_Q = compression stress concentration factor for main journal fillet

Note. β_Q is defined as the ratio of the maximum equivalent stress (von Mises) occurring in the fillet due to the radial force, to the nominal compressive stress related to the web cross-section.

β_T = torsional stress concentration factor for main journal fillet

Note. α_T and β_T are defined as the ratio of the maximum equivalent shear stress occurring in the fillets under torsional load, to the nominal torsional stress related to the axially bored crankpin or journal cross-section. See *Figure 2.3.7 Stress concentration factors in crankshaft fillets*.

γ_B = bending stress concentration factor for outlet of crankpin oil bore

γ_T = torsional stress concentration factor for outlet of crankpin oil bore

Note. γ_B and γ_T are defined as the ratio of the maximum principal stress occurring at the outlet of the crankpin oil-hole under bending and torsional loads, respectively, to the corresponding nominal stress related to the axially bored crankpin cross-section. See *Figure 2.3.8 Stress concentration factors and stress distribution at the edges of oil drillings*.

σ_{add} = additional bending stress due to misalignment and bedplate deformation as well as due to axial and bending vibrations

σ_B = specified minimum UTS of crankshaft material, in N/mm^2

σ_{BFN} = nominal alternating bending stress related to the web, in N/mm^2

σ_{BG} = alternating bending stress in journal fillet, in N/mm²

σ_{BH} = alternating bending stress in crankpin fillet, in N/mm²

σ_{BO} = alternating bending stress in the outlet of the oil bore, in N/mm²

σ_{BON} = nominal alternating bending stress in the outlet of the oil bore related to the crankpin diameter, in N/mm²

σ_{QFN} = nominal alternating compressive stress due to radial force related to the web, in N/mm²

σ_{DW} = allowable fatigue strength of crankshaft, in N/mm²

σ_{SP} = minimum yield strength of material for journal pin, in N/mm²

σ_{SW} = minimum yield strength of material for crankweb, in N/mm²

σ_{TO} = alternating torsional stress in the outlet of the crankpin oil bore, in N/mm²

σ_y = equivalent alternating stress for crankpin fillet, journal fillet or outlet of crankpin oil bore as applicable, in N/mm³

τ_H = alternating torsional stress in crankpin fillet, in N/mm²

τ_G = alternating torsional stress in journal fillet, in N/mm²

τ_N = calculated nominal alternating torsional stress referred to crankpin or journal (as applicable), in N/mm²

τ_a = manufacturer stated crankshaft half range torsional stress limit, in N/mm²

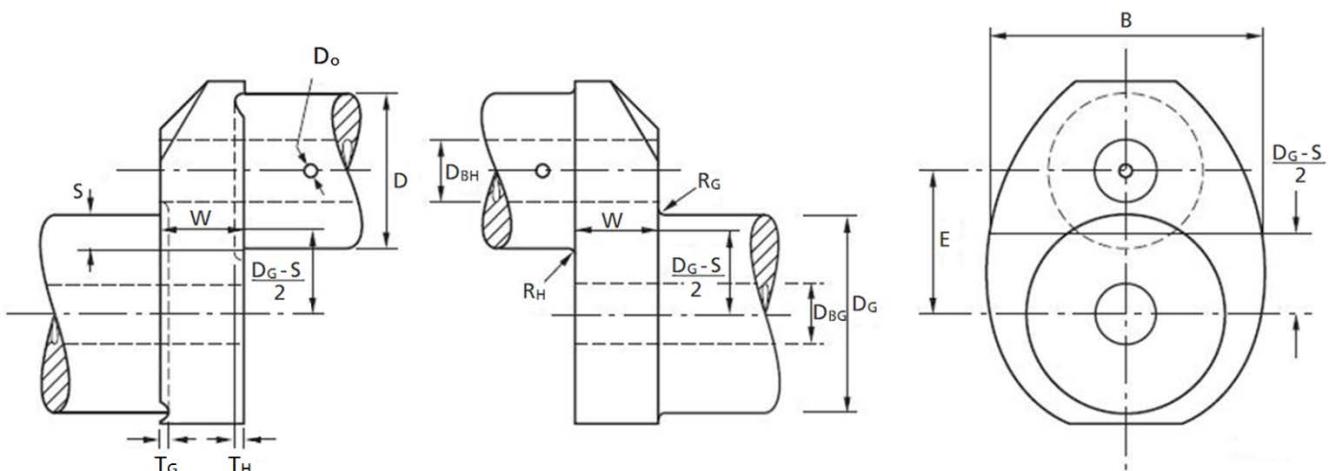
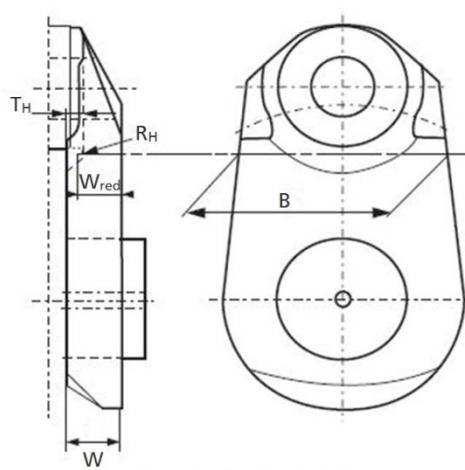


Figure 2.3.3 Crank dimensions for overlapped crankshaft



Crankshaft without overlap

Figure 2.3.4 Crank dimensions for crankshaft without overlap

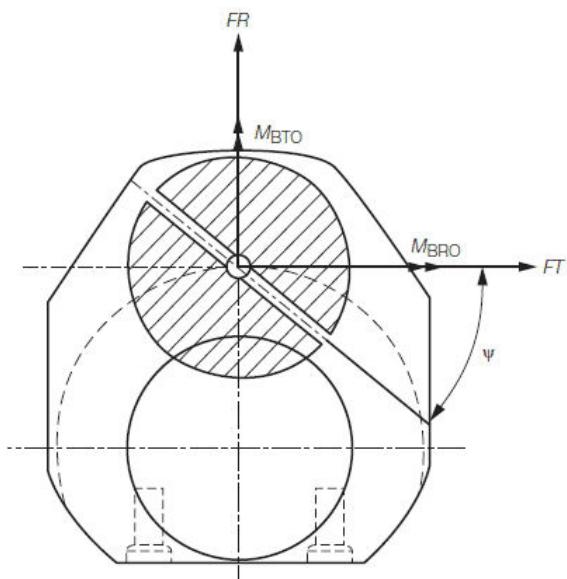


Figure 2.3.5 Crankpin section through the oil bore

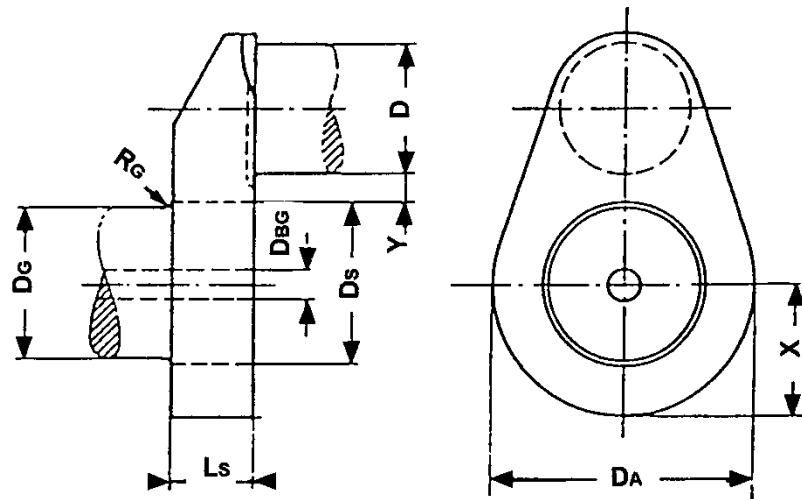
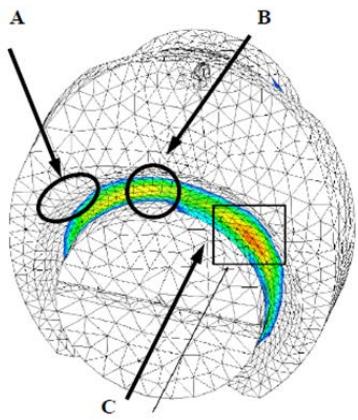
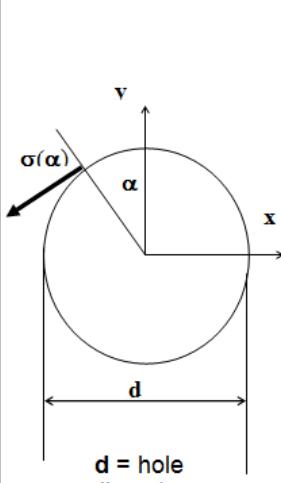


Figure 2.3.6 Crankthrow of semi-built crankshaft



Stress Type	Stress	Max $ \sigma_3 $	Max σ_1	
Torsional loading	Location of maximal stresses	<i>A</i>	<i>C</i>	<i>B</i>
	Typical principal stress system			
	Mohr's circle diagram with $\sigma_2 = 0$	$ \sigma_3 > \sigma_1$	$\sigma_1 > \sigma_3 $	$\sigma_1 \approx \sigma_3 $
Bending loading	Equivalent stress and S.C.F.	$\tau_{\text{equiv}} = \frac{\sigma_1 - \sigma_3}{2}$ $\text{S.C.F.} = \frac{\tau_{\text{equiv}}}{\tau_n} \text{ for } \alpha_T, \beta_T$		
	Location of maximal stresses	<i>B</i>	<i>B</i>	<i>B</i>
	Typical principal stress system			$\sigma_2 \neq 0$
Equivalent stress and S.C.F.	$\sigma_{\text{equiv}} = \sqrt{\sigma_1^2 + \sigma_2^2 - \sigma_1 \cdot \sigma_2}$ $\text{S.C.F.} = \frac{\sigma_{\text{equiv}}}{\sigma_n} \text{ for } \alpha_B, \beta_B, \beta_Q$			

Figure 2.3.7 Stress concentration factors in crankshaft fillets



Stress type	Nominal stress tensor	Uniaxial stress distribution around the edge	Mohr's circle diagram
Tension	$\begin{bmatrix} \sigma_n & 0 \\ 0 & 0 \end{bmatrix}$	$\sigma_\alpha = \sigma_n \gamma_B / 3 [1 + 2 \cos(2\alpha)]$	
Shear	$\begin{bmatrix} 0 & \tau_n \\ \tau_n & 0 \end{bmatrix}$	$\sigma_\alpha = \gamma_T \tau_n \sin(2\alpha)$	
Tension + shear	$\begin{bmatrix} \sigma_n & \tau_n \\ \tau_n & 0 \end{bmatrix}$	$\sigma_\alpha = \frac{\gamma_B}{3} \sigma_n \left\{ 1 + 2 \left[\cos(2\alpha) + \frac{3 \gamma_T \tau_n}{2 \gamma_B \sigma_n} \sin(2\alpha) \right] \right\}$	$\sigma_{\text{max}} = \frac{\gamma_B}{3} \sigma_n \left[1 + 2 \sqrt{1 + \frac{9}{4} \left(\frac{\gamma_T \tau_n}{\gamma_B \sigma_n} \right)^2} \right]$ $\text{for } \alpha = \frac{1}{2} \text{tg}^{-1} \left(\frac{3 \gamma_T \tau_n}{2 \gamma_B \sigma_n} \right)$

Figure 2.3.8 Stress concentration factors and stress distribution at the edges of oil drillings

3.5 Calculation of alternating stresses due to bending moments and radial forces – assumptions

3.5.1 The calculation is based on a statically determined system, composed of a single crankthrow supported in the centre of adjacent main journals and subject to gas and inertia forces. The bending length is taken as the length between the two main bearing midpoints (distance L_3 , see *Figure 2.3.9 Bending moment and shear force for in-line engine crankthrows* and *Figure 2.3.10 Bending moment and shear force for V engine crankthrows*).

3.5.2 The bending moments, M_{BR} and M_{BT} , are calculated in the relevant section based on triangular bending moment diagrams due to the radial component F_R and tangential component F_T of the connecting-rod force, respectively (see *Figure 2.3.9 Bending moment and shear force for in-line engine crankthrows*). For crankthrows with two connecting-rods acting upon one crankpin the relevant bending moments are obtained by superposition of the two triangular bending moment diagrams according to phase (see *Figure 2.3.9 Bending moment and shear force for V engine crankthrows*).

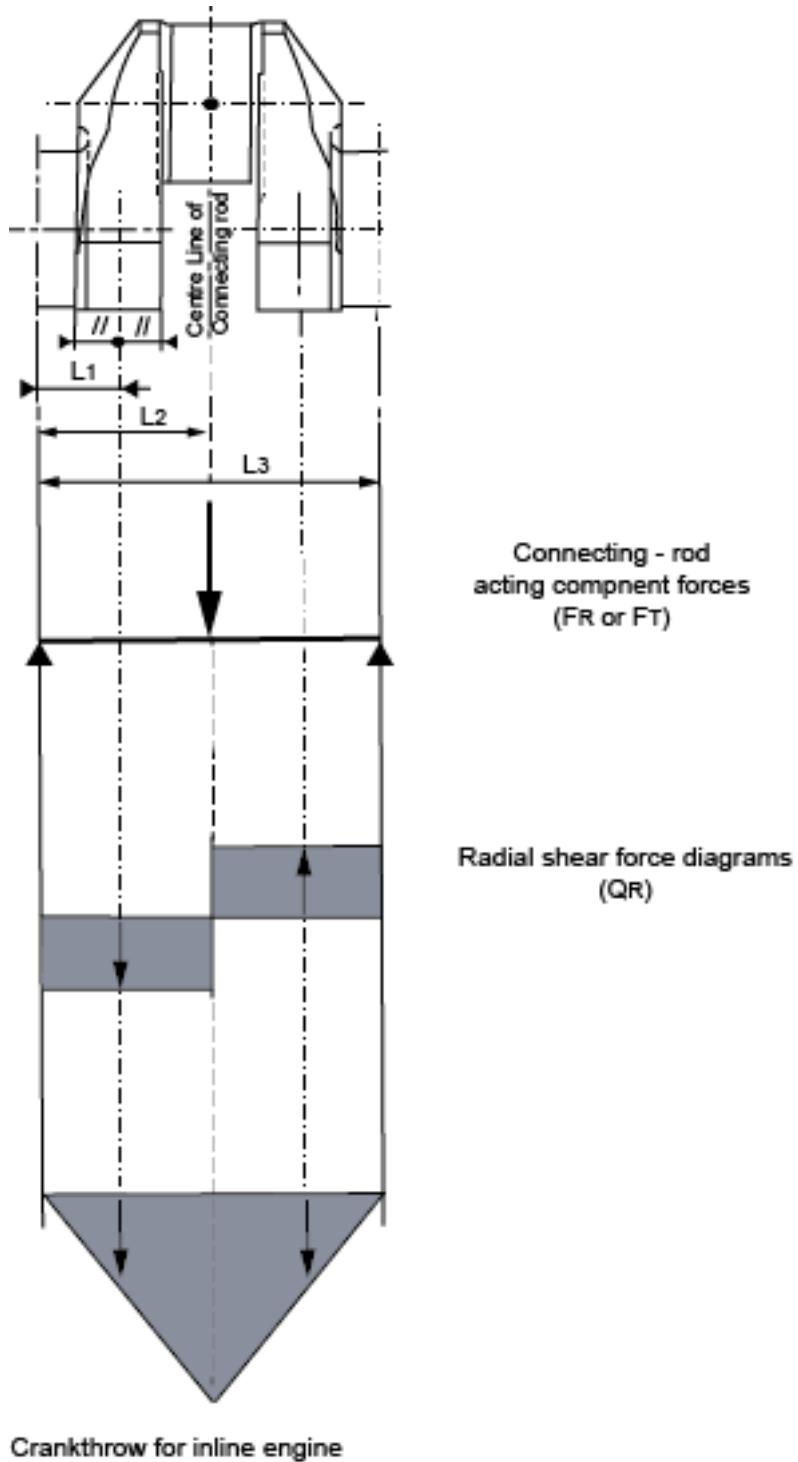


Figure 2.3.9 Bending moment and shear force for in-line engine crankthrows

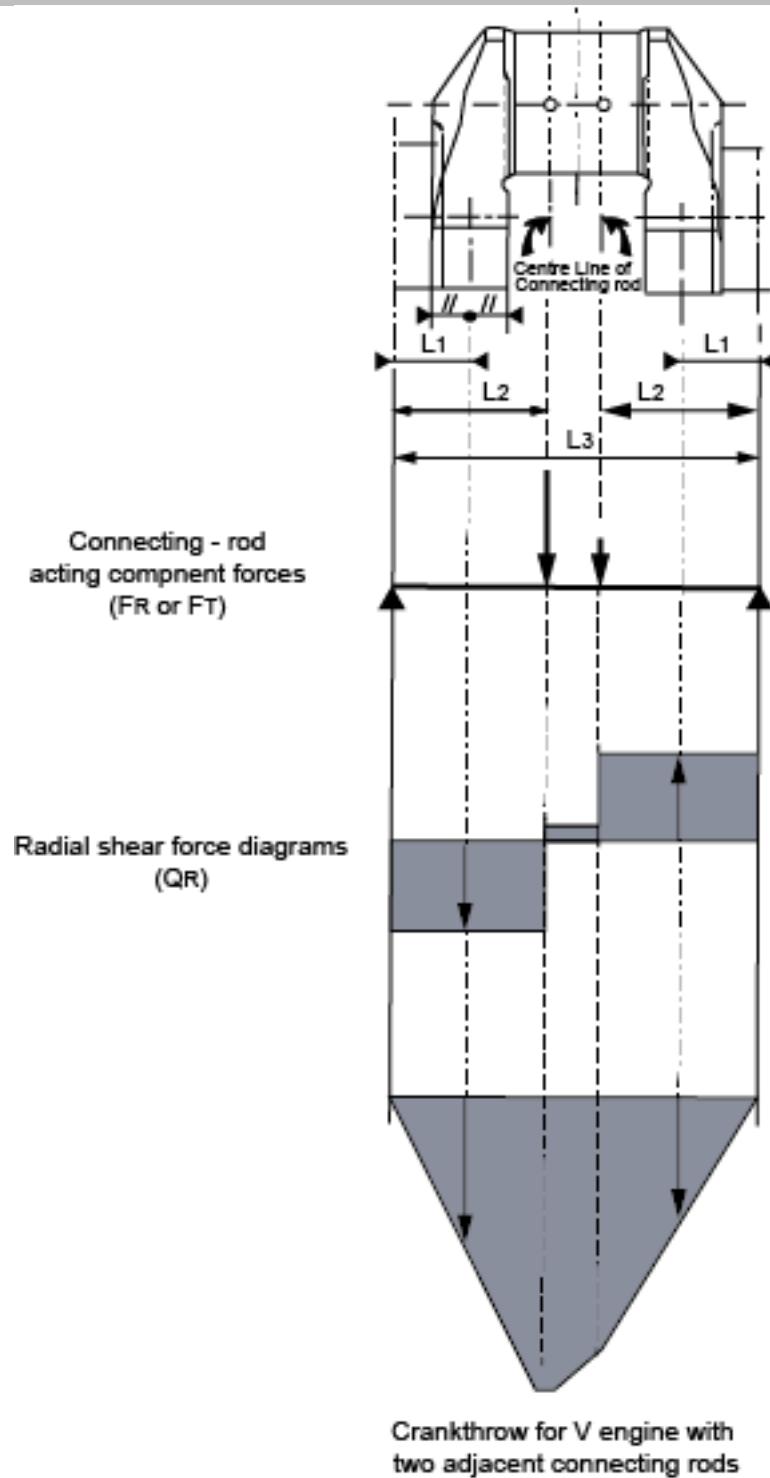


Figure 2.3.10 Bending moment and shear force for V engine crankthrows

3.5.3 The bending moment M_{BRF} and the radial force Q_{RF} are taken as acting in the centre of the solid web (distance L_1) and are derived from the radial component of the connecting-rod force. The alternating bending and compressive stresses due to bending moments and radial forces are to be related to the cross-section of the crankweb. This reference section results from the web thickness W and the web width B (see *Figure 2.3.3 Crank dimensions for overlapped crankshaft* and *Figure 2.3.4 Crank dimensions for crankshaft without overlap*). Mean stresses are neglected.

3.5.4 The two relevant bending moments for bending acting on the outlet of crankpin oil bores are taken in the crankpin cross-section through the oil bore. See *Figure 2.3.9 Bending moment and shear force for in-line engine crankthrows and Figure 2.3.10 Bending moment and shear force for V engine crankthrows*. M_{BRO} is the bending moment of the radial component of the connecting-rod force and M_{BTO} is the bending moment of the tangential component of the connecting-rod force. The alternating stresses due to these bending moments are to be related to the cross-sectional area of the axially bored crankpin. Mean bending stresses are neglected.

3.6 Calculation of bending stresses

3.6.1 The radial and tangential forces due to gas and inertia loads acting upon the crankpin at each connecting-rod position are to be calculated over one working cycle. Using the forces calculated over one working cycle and taking into account the distance from the main bearing midpoint, the time curve of the bending moments, M_{BRF} , M_{BRO} and M_{BTO} , and radial forces, Q_{RF} , as defined in *Pt 5, Ch 2, 3.5 Calculation of alternating stresses due to bending moments and radial forces – assumptions 3.5.2* and *Pt 5, Ch 2, 3.5 Calculation of alternating stresses due to bending moments and radial forces – assumptions 3.5.3* are then calculated.

3.6.2 Nominal bending stresses are referred to the web bending modulus.

3.6.3 In case of V-type engines, the bending moments – progressively calculated from the gas and inertia forces – of the two cylinders acting on one crankthrow are superposed according to phase. Different designs (forked connecting-rod, articulated-type connecting-rod or adjacent connecting-rods) shall be taken into account.

3.6.4 Where there are cranks of different geometrical configurations in one crankshaft, the calculation is to cover all crank variants.

3.6.5 The decisive alternating values will then be calculated according to:

$$X_N = \pm \frac{1}{2}(X_{\max} - X_{\min})$$

where

X_N is considered as alternating force, moment or stress

X_{\max} is maximum value within one working cycle

X_{\min} is minimum value within one working cycle

3.6.6 Nominal alternating bending and compressive stresses in web cross-section are calculated as follows:

$$\sigma_{BFN} = \pm \frac{M_{BRFN}}{W_{eqw}} \cdot 10^3 \cdot K_e \text{ N/mm}^2$$

$$\sigma_{QFN} = \pm \frac{Q_{RFN}}{F} \cdot K_e \text{ N/mm}^2$$

where

$$M_{BRFN} = \pm \frac{1}{2}(M_{BRF\max} - M_{BRF\min}) \text{ Nm}$$

$$W_{eqw} = \frac{BW^2}{6} \text{ mm}^3$$

K_e = 0,8 for crosshead engines
= 1,0 for trunk piston engines

$$Q_{RFN} = \pm \frac{1}{2}(Q_{RF\max} - Q_{RF\min}) \text{ N}$$

$$F = BW \text{ mm}^2$$

3.6.7 Nominal alternating bending stress in the outlet of the crankpin oil bore is calculated as follows:

$$\sigma_{BON} = \pm \frac{M_{BON}}{W_e} \cdot 10^3 \text{ N/mm}^2$$

where

M_{BON} is taken as the half range value $M_{BON} = 0,5(M_{BO\max} - M_{BO\min})$

and

$M_{BO} = (M_{BTO} \cos\psi + M_{BRO} \sin\psi)$, ψ = angular position in degrees, see *Figure 2.3.5 Crankpin section through the oil bore*

M_{BRO} = bending moment of the radial component of the connecting-rod force

M_{BTO} = bending moment of the tangential component of the connecting-rod force

$$W_e = \frac{\pi}{32} \left(\frac{D^4 - D_{BH}^4}{D} \right) \text{ mm}^3$$

3.6.8 Alternating bending stresses for the crankpin fillet and journal fillet are calculated as follows:

(a) For the crankpin fillet:

$$\sigma_{BH} = \pm(\alpha_B \sigma_{BFN}) \text{ N/mm}^2$$

where

α_B is calculated according to Pt 5, Ch 2, 3.8 Stress concentration factors 3.8.5 (a)

(b) For the journal fillet:

$$\sigma_{BG} = \pm(\beta_B \sigma_{BFN} + \beta_Q \sigma_{QFN}) \text{ N/mm}^2$$

where

β_B is calculated according to Pt 5, Ch 2, 3.8 Stress concentration factors 3.8.6 (a)

β_Q is calculated according to Pt 5, Ch 2, 3.8 Stress concentration factors 3.8.6 (b)

3.6.9 Alternating bending stresses for the outlet of crankpin oil bore are calculated as follows:

$$\sigma_{BO} = \pm(\gamma_B \sigma_{BON}) \text{ N/mm}^2$$

where

γ_B is calculated according to Pt 5, Ch 2, 3.8 Stress concentration factors 3.8.7 (a)

3.7 Calculation of torsional stresses

3.7.1 The nominal alternating torsional stress, τ_N , is to be taken into consideration. The value is to be derived from forced-damped vibration calculations of the complete dynamic system. Alternative methods will be given consideration. The engine designer is to advise the maximum level of alternating vibratory stress that is permitted (τ_a).

3.7.2 τ_a or τ_N (as applicable) is to be applied as a limiting value for the torsional vibration assessment required by Pt 5, Ch 8, 2 torsional vibration.

3.7.3 Nominal alternating torsional stress is calculated as follows:

$$\tau_N = \pm \frac{M_{TN}}{W_p} \cdot 10^3 \text{ N/mm}^2$$

where

$$M_{TN} = \pm \frac{1}{2} (M_{Tmax} - M_{Tmin}) \text{ Nm}$$

$$W_p = \frac{\pi}{16} \left(\frac{D^4 - D_{BH}^4}{D} \right) \text{ mm}^3 \text{ for the crankpin, or } W_p = \frac{\pi}{16} \left(\frac{D_G^4 - D_{BG}^4}{D_G} \right) \text{ mm}^3 \text{ for the journal}$$

τ_N is to be ascertained from assessment of the torsional vibration calculations where the maximum and minimum torques are determined for every mass point of the complete dynamic system and for the entire speed range by means of a harmonic synthesis of the forced vibrations from the first order up to and including the 15th order for 2-stroke cycle engines and from the 0,5th order up to and including the 12th order for 4-stroke cycle engines. While doing so, allowance must be made for the damping that exists in the system and for unfavourable conditions (misfiring in one of the cylinders when no combustion occurs but only compression cycle). The speed step calculation shall be selected in such a way that any resonance found in the operational speed range of the engine shall be detected.

3.7.4 For the purpose of the crankshaft assessment, the nominal alternating torsional stress considered in calculations is to be the highest calculated value, according to the method described in Pt 5, Ch 2, 3.7 Calculation of torsional stress 3.7.3, occurring at the most torsionally loaded mass point of the crankshaft system.

3.7.5 The approval of the crankshaft will be based on the installation having the largest nominal alternating torsional stress (but not exceeding the maximum figure specified by the engine manufacturer). For each installation it is to be ensured by calculation that the maximum approved nominal alternating torsional stress is not exceeded. See Pt 5, Ch 8, 2 Torsional vibration.

3.7.6 Alternating torsional stresses for the crankpin fillet, the journal fillet and the outlet of the crankpin oil bore are calculated as follows:

(a) Maximum alternating torsional stress in crankpin fillet:

$$\tau_H = \pm(\alpha_T \tau_N) \text{ N/mm}^2$$

where

α_T is calculated according to Pt 5, Ch 2, 3.8 Stress concentration factors 3.8.5 (b)

(b) Maximum alternating torsional stress in the journal fillet (not applicable to semi-built crankshafts):

$$\tau_G = \pm(\beta_T \tau_N) \text{ N/mm}^2$$

where

β_T is calculated according to Pt 5, Ch 2, 3.8 Stress concentration factors 3.8.6 (c)

(c) Maximum alternating torsional stress in the outlet of the crankpin oil bore:

$$\sigma_{TO} = \pm(\gamma_T \tau_N) \text{ N/mm}^2$$

where

γ_T is calculated according to Pt 5, Ch 2, 3.8 Stress concentration factors 3.8.7 (b).

3.8 Stress concentration factors

3.8.1 Stress concentration factors (SCF) are to be calculated using the analytical formulae outlined in this Section.

3.8.2 Crankshaft variables to be used in calculating the geometric stress concentrations factors are shown in *Table 2.3.1 Crankshaft variables for SCF calculation*, their limits of applicability are shown in *Table 2.3.2 Crankshaft variable boundaries for analytical SCF calculation*.

3.8.3 Where the geometry of the crankshaft is outside the boundaries (see *Table 2.3.2 Crankshaft variable boundaries for analytical SCF calculation*) of the analytical SCF, the calculation method detailed in Chapter 1 of *LR Guidance Notes for Crankshaft SCF Calculation using Finite Element Method* may be undertaken.

3.8.4 Where reliable experimental measurements and/or calculations are available, which can allow direct assessment of SCF, these can be used. The relevant documents and their analysis are to be submitted for consideration in order to demonstrate their equivalence. This is always to be performed when dimensions are outside the boundaries shown in *Table 2.3.2 Crankshaft variable boundaries for analytical SCF calculation*.

3.8.5 Chapters 1 and 3 of *LR Guidance Notes for Crankshaft SCF Calculation using Finite Element Method* describe how finite element (FE) analyses can be used for the calculation of the SCF. Care needs to be taken to avoid mixing equivalent (von Mises) stresses and principal stresses.

Table 2.3.1 Crankshaft variables for SCF calculation

Variable	Function
r	$= R_H/D$ for crankpin fillet $= R_G/D$ for journal fillet
s	$= S/D$
w	$= W/D$ crankshafts with overlap $= W_{\text{ref}}/D$ crankshafts without overlap
b	$= B/D$
d_o	$= D_o/D$
d_G	$= D_{BG}/D$
d_H	$= D_{BH}/D$
t_H	$= T_H/D$
t_G	$= T_G/D$

Table 2.3.2 Crankshaft variable boundaries for analytical SCF calculation

Lower bound	Variable	Upper bound
	s	$\leq 0,5$
0,2 \leq	w	$\leq 0,8$
1,1 \leq	b	$\leq 2,2$
0,03 \leq	r	$\leq 0,13$
0 \leq	d_G	$\leq 0,8$
0 \leq	d_H	$\leq 0,8$
0 \leq	d_o	$\leq 0,2$

Notes

Low range of s can be extended down to large negative values provided that:

- If calculated $f(\text{recess}) < 1$, then the factor $f(\text{recess})$ is not to be considered ($f(\text{recess}) = 1$)
- If $s < -0,5$, then $f(s, w)$ and $f(r, s)$ are to be evaluated replacing actual value of s by $-0,5$.

3.8.6 Crankpin SCF are calculated as follows:

(a) Bending

$$\alpha_B = 2,6914 f(s, w) \cdot f(w) \cdot f(b) \cdot f(r) \cdot f(d_G) \cdot f(d_H) \cdot f(\text{recess})$$

where

$$f(s, w) = -4,1883 + 29,2004w - 77,5925w^2 + 91,9454w^3 - 40,0416w^4 + (1-s)(9,5440 - 58,3480w + 159,3415w^2 - 192,5846w^3 + 85,2916w^4) + (1-s)^2(-3,8399 + 25,0444w - 70,5571w^2 + 87,0328w^3 - 39,1832w^4)$$

$$f(w) = 2,1790w^{0,7171}$$

$$f(b) = 0,684 - 0,0077b + 0,1473b^2$$

$$f(r) = 0,2081r^{(-0,5231)}$$

$$f(d_G) = 0,9993 + 0,27d_G - 1,0211d_G^2 + 0,5306d_G^3$$

$$f(d_H) = 0,9978 + 0,3145d_H - 1,5241d_H^2 + 2,4147d_H^3$$

$$f(\text{recess}) = 1 + (t_H + t_G)(1,8 + 3,2s)$$

(b) Torsion

$$\alpha_T = 0,8f(r, s) \cdot f(b) \cdot f(w)$$

where

$$f(r, s) = r^{(-0,322 + 0,1015(1-s))}$$

$$f(b) = 7,8955 - 10,654b + 5,3482b^2 - 0,857b^3$$

$$f(w) = w^{(-0,145)}$$

3.8.7 Journal fillet SCF are calculated as follows (not applicable to semi-built crankshafts):

(a) Bending

$$\beta_B = 2,7146 f_B(s, w) \cdot f_B(w) \cdot f_B(b) \cdot f_B(r) \cdot f_B(d_G) \cdot f_B(d_H) \cdot f(\text{recess})$$

where

$$f_B(s, w) = -1,7625 + 2,9821w - 1,5276w^2 + (1-s)(5,1169 - 5,8089w + 3,1391w^2) + (1-s)^2(-2,1567 + 2,3297w - 1,2952w^2)$$

$$f_B(w) = 2,2422w^{0,7548}$$

$$f_B(b) = 0,5616 + 0,1197b + 0,1176b^2$$

$$f_B(r) = 0,1908r^{(-0,5568)}$$

$$f_B(d_G) = 1,0012 - 0,6441d_G + 1,2265d_G^2$$

$$f_B(d_H) = 1,0022 - 0,1903d_H + 0,0073d_H^2$$

$$f(\text{recess}) = 1 + (t_H + t_G)(1,8 + 3,2s)$$

(b) Compression due to the radial force:

$$\beta_Q = 3,0128 f_Q(s) \cdot f_Q(w) \cdot f_Q(b) \cdot f_Q(r) \cdot f_Q(d_H) \cdot f(\text{recess})$$

where

$$f_Q(s) = 0,4368 + 2,1630(1-s) - 1,5212(1-s)^2$$

$$f_Q(w) = \frac{w}{0,0637 + 0,9369w}$$

$$f_Q(b) = b - 0,5$$

$$f_Q(r) = 0,5331r^{(-0,2038)}$$

$$f_Q(d_H) = 0,9937 - 1,1949d_H + 1,7373d_H^2$$

$$f(\text{recess}) = 1 + (t_H + t_G)(1,8 + 3,2s)$$

(c) Torsion:

$\beta_T = \alpha_T$ if the diameters and fillet radii of crankpin and journal are the same, or

$\beta_T = 0,8f(r, s) \cdot f(b) \cdot f(w)$ if crankpin and journal diameters and/or radii are of different sizes

where

$f(r, s)$, $f(b)$ and $f(w)$ are to be determined in accordance with Pt 5, Ch 2, 3.8 Stress concentration factors 3.8.6 (b), however, the radius of the journal fillet is to be related to the journal diameter: $r = \frac{R_G}{D_G}$

3.8.8 Crankpin oil bore SCF for radially drilled oil holes are calculated as follows:

(a) Bending

$$\gamma_B = 3 - 5,88d_o + 34,6d_o^2$$

(b) Torsion

$$\gamma_T = 4 - 6d_o + 30d_o^2$$

3.9 Additional bending stress

3.9.1 In addition to the alternating bending stresses in fillets (see Pt 5, Ch 2, 3.6 Calculation of bending stresses 3.6.8) further bending stresses due to misalignment and bedplate deformation as well as due to axial and bending vibrations are to be considered by applying σ_{add} as given by Table 2.3.3 Additional bending stresses.

Table 2.3.3 Additional bending stresses

Type of engine	σ_{add}
Crosshead engines	$\pm 30 \text{ N/mm}^2$ (see Note 1)
Trunk piston engines	$\pm 10 \text{ N/mm}^2$
Note 1. The additional stress of $\pm 30 \text{ N/mm}^2$ is composed of two components:	
(a) an additional stress of $\pm 20 \text{ N/mm}^2$ resulting from axial vibration	
(b) an additional stress of $\pm 10 \text{ N/mm}^2$ resulting from misalignment/bedplate deformation	

3.9.2 It is recommended that a value of $\pm 20 \text{ N/mm}^2$ be used for the axial vibration component for assessment purposes where axial vibration calculation results of the complete dynamic system (engine/shafting/gearing/propeller) are not available. Where axial vibration calculation results of the complete dynamic system are available, the calculated figures can be used instead.

3.10 Equivalent alternating stress

3.10.1 In the fillets, bending and torsion lead to two different biaxial stress fields which can be represented by a von Mises equivalent stress with the additional assumptions that bending and torsion stresses are time phased and the corresponding peak values occur at the same location (see Figure 2.3.7 Stress concentration factors in crankshaft fillets). As a result the equivalent alternating stress is to be calculated for the crankpin fillet as well as for the journal fillet by using the von Mises criterion.

3.10.2 At the oil hole outlet, bending and torsion lead to two different stress fields which can be represented by an equivalent principal stress equal to the maximum of principal stress resulting from combination of these two stress fields with the assumption that bending and torsion are time phased (see Figure 2.3.8 Stress concentration factors and stress distribution at the edges of oil drillings).

3.10.3 The above two different ways of equivalent stress evaluation both lead to stresses which can be compared to the same fatigue strength value of crankshaft assessed according to the von Mises criterion.

3.10.4 Equivalent alternating stress, σ_v , is defined as:

(a) For the crankpin fillet:

$$\sigma_v = \pm \sqrt{(\sigma_{BG} + \sigma_{\text{add}})^2 + 3\tau_G^2} \text{ N/mm}^2$$

(b) For the journal fillet:

$$\sigma_v = \pm \sqrt{(\sigma_{BH} + \sigma_{\text{add}})^2 + 3\tau_H^2} \text{ N/mm}^2$$

(c) For the outlet of crankpin oil bore:

$$\sigma_v = \pm \frac{1}{3} \sigma_{BO} \left[1 + 2 \sqrt{1 + \frac{9}{4} \left(\frac{\sigma_{TO}}{\sigma_{BO}} \right)^2} \right] \text{ N/mm}^2$$

3.11 Fatigue strength

3.11.1 The fatigue strength is to be understood as that value of equivalent alternating stress (von Mises) which a crankshaft can permanently withstand at the most highly stressed points. The fatigue strength can be evaluated by means of the following formulae.

(a) Related to the crankpin diameter:

$$\sigma_{DW} = \pm K(0,42\sigma_B + 39,3) \left[0,264 + 1,073D^{-0.2} + \frac{785-\sigma_B}{4900} + \frac{196}{\sigma_B} \sqrt{\frac{1}{R_X}} \right] \text{ N/mm}^2$$

with

$R_X = R_H$ in the fillet area

$R_X = D_o/2$ in the oil bore area

(b) Related to the journal diameter:

$$\sigma_{DW} = \pm K(0,42\sigma_B + 39,3) \left[0,264 + 1,073D_G^{-0.2} + \frac{785-\sigma_B}{4900} + \frac{196}{\sigma_B} \sqrt{\frac{1}{R_G}} \right] \text{ N/mm}^2$$

where

$$K = K_1 K_2$$

K_1 = fatigue endurance factor appropriate to the manufacturing process

= 1,05 for continuous grain flow forged or drop-forged crankshafts

= 1,0 for free form forged crankshafts (without continuous grain flow)

= 0,93 for cast steel crankshafts with cold rolling treatment in fillet area manufactured by companies using an LR approved cold rolling process

K_2 = fatigue enhancement factor for surface treatment. These treatments are to be applied to the fillet radii

A value for K_2 will be assigned upon application by the engine designers. Full details of the process, together with the results of full scale fatigue tests will be required to be submitted for consideration. See *LR Guidance note - Guidance for the evaluation of Crankshaft Fatigue Tests*. Alternatively, the following values may be taken (surface hardened zone to include fillet radii):

$$K_2 \begin{aligned} &= 1,15 \text{ for induction hardened} \\ &= 1,25 \text{ for nitrided} \end{aligned}$$

Where a value of K_1 or K_2 greater than unity is to be applied then details of the manufacturing process are to be submitted. An enhanced K_1 factor will be considered, subject to special approval of the manufacturing specification. See *Materials and Qualification Procedures for Ships, Book E, Procedure MQPS 5-2*.

3.11.2 The formulae in *Pt 5, Ch 2, 3.11 Fatigue strength 3.11.1* are subject to geometry limits. The junction of the oil hole with the crankpin or main journal surface is to be formed with an adequate radius and smooth surface finish down to a minimum depth equal to 1,5 times the oil bore diameter and for calculation purposes R_H , R_G or R_X are to be taken as not less than 2 mm.

3.11.3 Fatigue strength calculations or alternatively fatigue test results determined by experiment based either on full size crankthrow (or crankshaft) or on specimens taken from a full size crankthrow may be required to demonstrate acceptability. The experimental procedure for fatigue evaluation of specimens and fatigue strength of crankshaft assessment are to be submitted for approval by LR. The procedure is to include as a minimum: method, type of specimens, number of specimens (or crankthrows), number of tests, survival probability, and confidence number. See also *LR Guidance for the Evaluation of Crankshaft Fatigue Tests*.

3.11.4 When journal diameter is equal or larger than the crankpin diameter, the outlets of main journal oil bores are to be formed in a similar way to the crankpin oil bores, otherwise separate fatigue strength calculations or, alternatively, fatigue test results may be required.

3.11.5 Only surface treatment processes approved by LR are permitted. Guidance for calculation of surface treated fillets and oil bore outlets is presented in Chapter 2 of the *LR Guidance Notes for Crankshaft SCF Calculation using Finite Element Method*.

3.12 Acceptability criteria

3.12.1 The sufficient dimensioning of a crankshaft is confirmed by a comparison of the equivalent alternating stress and the fatigue strength. The acceptability factor, Q , is to be greater than or equal to 1,15 for the crankpin fillet, the journal fillet, the outlet of crankpin oil bore:

$$Q = \frac{\sigma_{DW}}{\sigma_v}$$

3.13 Shrink fit of semi-built crankshafts

3.13.1 The following formulae are applicable to crankshafts assembled by shrinking main journals into the crankwebs, see also *Figure 2.3.6 Crankthrow of semi-built crankshaft*.

3.13.2 In general, the radius of transition, R_G , between the main journal diameter, D_G , and the shrink diameter, D_S , is to be not less than $0,015D_G$ or $0,5(D_S - D_G)$ where the greater value is to be considered.

3.13.3 Deviations from these parameters will be specially considered.

3.13.4 The maximum permissible internal diameter in the journal pin is to be calculated in accordance with the following formula, this condition serves to avoid plasticity in the hole of the journal pin:

$$D_{BG} = D_S \sqrt{1 - \frac{4000S_R M_{max}}{\mu\pi D_S^2 L_S \sigma_{SP}}} \text{ mm}$$

where

S_R = safety factor against slipping; however, a value not less than 2 is to be taken unless documented by experiments.

M_{max} = absolute maximum value of the torque M_{Tmax} in accordance with *Pt 5, Ch 2, 3.6 Calculation of torsional stresses 3.6.3*, in N m

μ = coefficient for static friction; however, a value not greater than 0,2 is to be taken unless documented by experiments.

3.13.5 The actual oversize Z of the shrink fit must be within the limits Z_{min} and Z_{max} calculated in accordance *Pt 5, Ch 2, 3.13 Shrink fit of semi-built crankshafts 3.13.7* and *Pt 5, Ch 2, 3.13 Shrink fit of semi-built crankshafts 3.13.8*. When *Pt 5, Ch 2, 3.13 Shrink fit of semi-built crankshafts 3.13.5* cannot be complied with, then the calculated values of Z_{min} and Z_{max} are not applicable due to multizone-plasticity problems. In such cases Z_{min} and Z_{max} are to be established from FEM calculations.

3.13.6 The minimum required diametral Interference is to be taken as the greater of:

$$Z_{min} \geq \frac{\sigma_{sw} D_S}{E_m} \text{ mm}$$

and

$$Z_{\min} \geq \frac{4000}{\mu\pi} \frac{S_R M_{\max}}{E_m D_S L_S} \frac{1 - Q_A^2 Q_S^2}{(1 - Q_A^2)(1 - Q_S^2)} \text{ mm}$$

where

$$Q_A = \text{web ratio, } Q_A = \frac{D_S}{D_A}$$

$$Q_S = \text{shaft ratio, } Q_S = \frac{D_{BG}}{D_S}$$

3.13.7 The maximum diametral interference is not to be greater than:

$$Z_{\max} \leq D_S \left(\frac{\sigma_{SW}}{E_m} + \frac{0.8}{1000} \right) \text{ mm}$$

This condition serves to restrict the shrinkage induced mean stress in the fillet.

3.13.8 Reference marks are to be provided on the outer junction of the crankwebs with the journals.

■ **Section 4** **Electronically controlled engines**

4.2 Risk-based analysis

4.2.3 A risk-based analysis is to be carried out for:

- (c) main engines on ships with multiple main engines or other means of providing propulsion power; and/or
- (d) auxiliary engines intended to drive electric generators forming the ship's main source of electrical power or otherwise providing power for essential services.

The analysis is to demonstrate that adequate hazard mitigation has been incorporated in electronically controlled engine systems or the overall ship installation with respect to personnel safety and providing propulsion power and/or power for essential services for the safety of the ship. Arrangements satisfying the criteria of *Pt 5, Ch 2, 4.2 Risk-based analysis 4.2.2 to Pt 5, Ch 2, 4.2 Risk-based analysis 4.2.2.(c)* will also be acceptable.

(Part only shown)

4.2.5 The risk-based analysis report is to:

- (d) Identify the equipment, system or sub-system; and the mode of operation and the equipment.

■ **Section 7** **Control and monitoring of main, auxiliary and emergency engines**

7.3 Auxiliary engine governors

7.3.2 If an engine cannot achieve the requirements of *Pt 5, Ch 2, 7.3 Auxiliary engine governors 7.3.1* then the actual load step is to be declared and verified through testing to ensure the requirements specified in *Pt 6, Ch 2, 1.8 Quality of power supplies* are satisfied. In cases where a step load equivalent to the rated output of a generator is switched off, a transient speed variation in excess of 10 per cent of the rated speed is acceptable, provided this does not cause the intervention of the overspeed device as required by *Pt 5, Ch 2, 7.4 Overspeed protective devices 7.4.1*.

(Part only shown)

7.3.3 Emergency engines are to comply with *Pt 5, Ch 2, 7.3 Auxiliary engine governors 7.3.1* except that the initial load required by *Pt 5, Ch 2, 7.3 Auxiliary engine governors 7.3.1.(b)* is to be not less than the total connected emergency statutory load, or if their total consumer load is applied in steps, the following requirements are to be met:

(Part only shown)

7.3.5 For alternating current installations, the permanent speed variation of the machines intended for parallel operation are to be equal within a tolerance of ± 0.5 per cent. Momentary speed variations with load changes in accordance with *Pt 10, Ch 1, 7.3 Auxiliary engine governors 7.3.1* are to return to and remain within one per cent of the final steady state speed. This should normally be accomplished within five but in no case more than eight seconds. For quality of power supplies, see *Pt 16, Ch 2, 1.7 Design and construction Pt 6, Ch 2, 1.8 Quality of power supplies*.

7.8 Emergency engines

7.8.1 Alarms and safeguards are to be fitted in accordance with indicated in *Table 2.7.4 Emergency engines: Alarms and safeguards*.

(Part only shown)

Table 2.7.4 Emergency engines: Alarms and safeguards

Item	Alarm for engine power <220 kW	Alarm for engine power ≥220kW	Note
Fuel oil leakage from pressure pipes	Leakage	Leakage	See Pt 5, Ch 2, 7.1 General 7.1.27.5 Unattended machinery 7.5.5

■ **Section 8 Piping**

8.1 Fuel oil, hydraulic and high pressure oil systems

Paragraphs 8.1.4 and 8.1.11 have been moved to new sub-Section 8.2. Existing paragraphs 8.1.5 to 8.1.10 have been renumbered 8.1.4 to 8.1.9.

(Part only shown)

8.1.12 8.1.10 For high-pressure oil containing and mechanical power transmission systems, the quality plan for sourcing, design, installation and testing of components is to address the following issues (see *Pt 5, Ch 2, 1.2 Submission requirements 1.2.1 Table 1.2.1 Plans and particulars to be submitted*, Note 11):

8.2 Additional requirements for fuel oil, hydraulic and high pressure oil systems for ships

8.1.4 8.2.1 Ships Vessels of less than 500 gross tons and which are not required to comply with the SOLAS - *International Convention for the Safety of Life at Sea*, are to be capable of maintaining adequate manoeuvring capability.

8.1.14 8.2.2 Where multi-engined installations are supplied from the same fuel source, means of isolating the fuel supply and spill piping to individual engines is to be provided. These means of isolation are not to affect the operation of the other engines and are to be operable from a position not rendered inaccessible by a fire on any of the engines.

Existing sub-Sections 8.2 and 8.3 have been renumbered 8.3 and 8.4.

■ **Section 9 Starting arrangements**

9.4 Additional requirements for electric starting for non-SOLAS cargo vessels

9.3.6 9.4.1 For cargo ships vessels of less than 500 gross tons which are not required to comply with the *International Convention for the Safety of Life at Sea, 1974*, as amended (SOLAS 74), the emergency source of electrical power may be used as one of the sources of energy required by *Pt 5, Ch 2, 9.3 Electric starting 9.3.1* or *Pt 5, Ch 2, 9.3 Electric starting 9.3.2* for electric starting. Where the emergency source of electrical power is an accumulator battery and it is to be used for electric starting, it is to have the additional capacity required to ensure emergency supplies are not compromised and is to be adequately protected and suitably located for use in an emergency.

Existing sub-sections 9.4 and 9.5 have been renumbered 9.5 and 9.6.

■ **Section 11 Factory Acceptance Test and Shipboard Trials of Engines**

11.3 Works trials (factory acceptance test)

(Part only shown)

Table 2.11.1 Scope of works trials for engines

Main engines driving propellers and waterjets		
Testing of governor and independent overspeed protective device	—	See <i>Pt 5, Ch 2, 7.2 Main engine governors 7 Control and monitoring of main, auxiliary and emergency engines</i>
Shutdown device	—	See <i>Pt 5, Ch 2, 7.4 Overspeed protective devices 7 Control and monitoring of main, auxiliary and emergency engines</i>
Engines driving mechanical auxiliaries		
Trial condition	Duration	Note

11.4 Shipboard trials

(Part only shown)

Table 2.11.2 Scope of shipboard trials for engines

Main engines driving fixed-pitch propeller or waterjet (see Note 1)		
Trial condition	Duration	Note
Starting and reversing manoeuvres	–	See Pt 5, Ch 2, 9 Starting arrangements and Pt 5, Ch 2, 13 Air compressors (see Note 5)
Note 1. For main propulsion engines driving reversing gears, the tests for main engines driving fixed-pitch propellers apply as appropriate.		
Note 5. Starting manoeuvres are to be carried out in order to verify the capacity of the starting media. The ability of reversible engines to be operated in the reverse direction is to be demonstrated. See Pt 5, Ch 1, 3.9 Astern power.		

11.4.8 In addition to the tests listed in *Table 2.11.2 Scope of shipboard trials for engines* other tests may also be required by statutory regulations (e.g. The engine is to be checked for stable running (steady fuel index) at both upper and lower borders of the barred speed range. Steady fuel index means an oscillation range less than five per cent of the effective stroke (idle to full index).

11.4.9 In addition to the tests listed in *Table 1.11.2 Scope of on board trials for diesel engines*, other tests may also be required by statutory regulations (e.g. the testing of exhaust gas emissions is to comply with *MARPOL - International Convention for the Prevention of Pollution from Ships* as applicable).

Part 5, Chapter 24

Emissions Abatement Plant for Combustion Machinery

■ Section 9

Electrical and control equipment

9.1 General

9.1.6 An emergency stop function is to be provided, which is to:

- Close quick-closing valves on chemical tank(s) (where applicable, *).
- Stop chemical feed pump(s) (where applicable*).
- Where fitted, open exhaust gas cleaning by-pass valve.
- Stop scrubber water pumps and close scrubber water inlet valve (where applicable).

Note. * Not required for emissions abatement plant utilising urea solution, see also Pt 5, Ch 24, 10 Storage and use of chemicals. Systems using other media are to be considered on a case-by-case basis.

Part 6, Chapter 2 Electrical Engineering

■ Section 1 General requirements

1.3 Documentation required for supporting evidence

1.3.9 Evidence demonstrating the compatibility of the converter, cable and motor combinations to be used for the provision of essential services. Particular attention is to be given the suitability of the insulation systems used with respect to the convertor impulse magnitudes and voltage rise times, and their implications for partial discharge.

1.3.10 For high voltage a.c. rotating machines rated at above 3,6 kV, an inspection and test plan is required which enables the requirements of *Pt 6, Ch 2, 9 Rotating Machines, 9.8.7* to be assessed.

1.4 Surveys

1.4.6 Alternative approach for product assurance:

- (a) LR will be prepared to give consideration to the adoption of an approach for product assurance, utilising regular and systematic audits of an organisation's arrangements for assuring product quality, as an alternative to the direct survey of individual items.
- (b) Alternative approaches for product assurance are to be approved by LR. In order to obtain approval, the requirements of *Pt 5, Ch 1, 6 Quality Assurance Scheme for Machinery* or *Ch 1, 2.4 Materials Quality Scheme of the Rules for the Manufacture, Testing and Certification of Materials, July 2017, incorporating Notice No. 1 & 2* are to be complied with. Proposals for equivalent approaches are to be submitted for consideration.

■ Section 9 Rotating machines

9.1 General requirements

9.1.4 All machines of 100 kW and over, intended for essential services, are to be surveyed by the Surveyor during manufacture and test, *see also Pt 6, Ch 2, 1.4 Surveys 1.4.6*.

9.8 Survey and testing

9.8.7 The partial discharge characteristics of the high voltage a.c. rotating machines for essential services rated at above 3,6 kV are to be measured and recorded in accordance with *Pt 6, Ch 2, 21.4 Partial discharge testing of high voltage rotating machines for essential services*.

■ Section 11 Electric cables, optical fibre cables and busbar trunking systems (busways)

11.1 General

11.1.4 Electric cables for electric propulsion systems are to be Type Approved in accordance with LR's *Type Approval System Test Specification Number 3* or, alternatively, surveyed by the Surveyors during manufacture and testing to assess compliance with the applicable International or National Standards and application of an acceptable quality management system, *see also Pt 6, Ch 2, 1.4 Surveys 1.4.6*.

■ Section 19 Ship safety systems

19.1 Watertight doors

19.1.1 Power operated sliding watertight doors including power supply, power supply availability, control, indication and alarm circuits for passenger ships are to comply with SOLAS Ch II-1, Regulation 13.7.1 to 13.7.8, ~~7 Additional requirements for ro-ro passenger ships~~.

19.1.2 The enclosures of electrical components including their electric control cables for passenger ships are to be in compliance with SOLAS Ch II-1, Regulation 13.7.6, ~~5 Means of escape on passenger ships from special category and open ro-ro spaces to which any passengers carried can have access~~.

19.1.3 For passenger ships, an audible alarm and where required supplemented by a visual signal at the door when the watertight doors are operated from remote is to be in accordance with SOLAS Ch II-1, Regulation 13.7.1.6, *7 Additional requirements for ro-ro passenger ships*.

19.1.4 The requirement of sliding watertight doors on cargo ships is to comply with SOLAS Ch II-1, Regulation 13.1, *2 General requirements*. For the necessity of a centralised operating console located on the navigation bridge on passenger vessels, it is to comply with SOLAS Ch II-1, Regulation 13.8.1 to 13.8.3.

19.1.5 For the necessity of a centralised operating console located on the navigation bridge on passenger vessels, it is to comply with SOLAS Ch II-1, *Regulation 13 - Means of escape* *Section 8*. The sliding watertight doors on cargo ships are to comply with the following requirements:

- (a) SOLAS Ch II-1, Regulation 13.1 - Openings in watertight bulkheads and internal decks in cargo ships.
- (b) Provisions are to be made as follows:
 - (i) The electrical power required for power-operated sliding watertight doors is to be separate from any other power circuit and supplied from the emergency switchboard either directly or by a dedicated distribution board situated above the bulkhead deck. The associated control, indication and alarm circuits are to be supplied from the emergency switchboard either directly or by a dedicated distribution board situated above the bulkhead deck.
 - (ii) A single failure in the power operating or control system of power-operated sliding watertight doors is not to result in a closed door opening or preventing the hand operation of any door.
 - (iii) Availability of the power supply is to be continuously monitored at a point in the electrical circuit adjacent to the door operating equipment. Loss of any such power supply is to activate an audible and visual alarm at the central operating console at the navigating bridge.
 - (iv) Electrical power, control, indication and alarm circuits are to be protected against fault in such a way that a failure in one door circuit will not cause a failure in any other door circuit. Short-circuits or other faults in the alarm or indicator circuits of a door are not to result in a loss of power operation of the door. Arrangements are to be such that leakage of water into the electrical equipment located below the bulkhead deck will not cause the door to open.
 - (v) The enclosures of electrical components necessarily situated below the bulkhead deck are to provide suitable protection against the ingress of water with ratings as defined in IEC 60529: Degrees of protection provided by enclosures (IP Code) or an acceptable and relevant National Standard, are as follows:
 - 1) Electrical motors, associated circuits and control components, protected to IPX7 Standard.
 - 2) Door position indicators and associated circuit components protected to IPX8 Standard, where the water pressure testing of the enclosures is to be based on the pressure that can occur at the location of the component during flooding for a period of 36 hours.
 - 3) Door movement warning signals, protected to IPX6 Standard.
 - (vi) Watertight door electrical controls including their electric cables are to be kept as close as is practicable to the bulkhead in which the doors are fitted and so arranged that the likelihood of them being involved in any damage which the ship can sustain is minimised.
 - (vii) An audible alarm, distinct from any other alarm in the area, is to sound whenever the door is closed remotely by power and sound for at least five seconds but no more than ten seconds before the door begins to move and is to continue sounding until the door is completely closed. The audible alarm is to be supplemented by an intermittent visual signal at the door and in areas where the noise level exceeds 85 dB(A).
 - (viii) Sliding watertight doors on cargo ships are to be capable of being remotely closed from the bridge and are also to be operable locally from each side of the bulkhead. Indicators are to be provided at the control position showing whether the doors are open or closed, and an audible alarm is to be provided at the door closure.

Part 7, Chapter 11

Arrangements and Equipment for Environmental Protection (ECO Class Notation)

■ **Section 1** **General requirements**

1.2 ECO class notation: minimum requirements and additional characters

(Part only shown)

1.2.2 Pt 7, Ch 11, 3 *Supplementary characters* contains additional requirements. Ships complying with these requirements will be eligible for one or more of the following associated supplementary characters, as applicable:

IHM Inventory of hazardous materials.

NO_{x1}, NO_{x2}, NO_{x3} Nitrogen Oxides (NO_x) exhaust emissions.

1.5 Information to be submitted

(Part only shown)

1.5.4 The following operational procedures are to be in place at the time of the on board verification survey:

~~(m) Procedures for maintaining inventory of hazardous materials (supplementary character **IHM** only).~~

(Part only shown)

1.5.5 Information and plans:

~~(w) Inventory of Hazardous Materials Statement of Compliance.~~

Items x to ac have been renumbered w to ab.

■ **Section 2** **Minimum requirements**

2.1 General

2.1.3 Where a ship, by virtue of its gross tonnage, is not required by the Antifouling Convention to have certification, an antifouling system (AFS) declaration in the format shown in Appendix 2 of Annex 4 to the Convention is to be maintained on board. The application of antifouling systems containing TBT above the level specified in the Antifouling Convention is prohibited.

2.1.4 Where a ship, by virtue of its gross tonnage, is not required to have a Safety Management Certificate (SMC), it is exempt from Pt 7, Ch 11, 2.1 General 2.1.2.(b).

2.1.5 High speed craft, as defined in LR's *Rules and Regulations for the Classification of Special Service Craft*, and non-tank ships, other than tankers, of less than 400 gross tonnes tonnage, will be the subject of special consideration.

2.1.6 Offshore supply vessels that are less than 100 m in length, as per MSC 235(82), are exempt from the requirement to be enrolled in LR's Ship Emergency Response Service (SERS) or the equivalent scheme of another IACS member. Exempting the requirement for enrolment in SERS, or the equivalent scheme of another IACS member for other vessel types, will be specially considered depending on the vessel's size and operational profile.

2.4 Energy management

2.4.1 A Ship Energy Efficiency Management Plan (SEEMP) Part I is to be retained on board according to the provisions of Regulation 22 - *Ship Energy Efficiency Management Plan (SEEMP)* of MARPOL Annex VI - *Regulations for the Prevention of Air Pollution from Ships* and subsequently be reviewed by LR.

~~2.4.2 The SEEMP is to be developed in accordance with Resolution MEPC.213(63) – 2012 Guidelines for the Development of a Ship Energy Efficiency Management Plan (SEEMP) – (Adopted on 2 March 2012).~~

2.4.2 For ships of 5000 gross tonnage and above, a SEEMP Part II is also to be retained on board. This is to be examined by LR for compliance with regulation 22.2 of MARPOL Annex VI and a SEEMP Part II Examination Page (LR Form 3202) is to be completed, inserted into the Plan and retained on board.

2.4.3 The SEEMP Part I and Part II are to meet the requirements of Resolution MEPC.282(70) – 2016 Guidelines for the Development of a Ship Energy Efficiency Management Plan (SEEMP) – (Adopted on 28 October 2016).

2.5 Refrigeration systems

2.5.1 These requirements apply to all permanently installed refrigeration and air conditioning installations on board with more than 3 kg of refrigerant. These requirements do not apply to stand-alone refrigerators, freezers and ice makers used in galleys, pantries, bars and crew accommodation.

Table 11.2.1 Refrigerant leak testing - maximum periodicity

Charge size	Periodicity	Leakage
under 3 kg	6 months	10%
3–30 kg	3 months	10%
30–300 kg	Monthly	5%
Over 300 kg	Monthly	<3%

2.10 Sewage treatment

2.10.1 Where fitted, the sewage treatment system is to be approved in accordance with ~~MEPC Resolution 159(55)~~ Resolution MEPC.284.(70). As an alternative, a Declaration of Conformity issued under the EU Marine Equipment Directive is acceptable.

Existing sub-Section 2.11 has been deleted in its entirety.

Existing sub-Sections 2.12 and 2.13 have been renumbered 2.11 and 2.12.

■ Section 3 Supplementary characters

3.7 Energy management – SEEMP and EnMS characters

3.7.1 For assignment of the **SEEMP** character, and in addition to the requirements specified in Pt 7, Ch 11, 2.4 *Energy management*, ~~a the SEEMP Part I~~ is to be reviewed by LR to check that it is in alignment with ~~the IMO Resolution MEPC.213(63)–2012 Guidelines for the Development of a Ship Energy Efficiency Management Plan (SEEMP) (Adopted on 2 March 2012)~~ ~~Resolution MEPC.282(70) – 2016 Guidelines for the Development of a Ship Energy Efficiency Management Plan (SEEMP) – (Adopted on 28 October 2016)~~ and reflects industry guidelines as applicable. This is to be demonstrated by the LR SEEMP Statement of Conformance and associated documentation.

3.7.2 For ships of 5000 gross tonnage and above, in addition to Pt 7, Ch 11, 3.7 *Energy management – SEEMP and EnMS characters* 3.7.1, a SEEMP Part II is to be examined by LR to check that it is in accordance with the ~~IMO Resolution MEPC.282(70) – 2016 Guidelines for the Development of a Ship Energy Efficiency Management Plan (SEEMP) – (Adopted on 28 October 2016)~~. This is to be demonstrated by the confirmation of compliance (LR SEEMP Part II examination cover page and the associated design appraisal document).

Existing paragraph 3.7.2 has been renumbered 3.7.3.

Existing sub-Section 3.10 has been deleted in its entirety.

3.11 3.10 Nitrogen oxides NO_{-x1} , NO_{-x2} , NO_{-x3} characters

3.11.4 3.10.1 For assignment of the NO_{-x1} -or NO_{-x2} -character, the total weighted value of NO_x emissions from all installed engines defined within Pt 7, Ch 11, 2.2 *Nitrogen oxides (NO_x)* 2.2.1 is not to exceed 80 per cent of the total weighted Tier II NO_x emission limits specified in MARPOL Annex VI, Regulation 13 – *Nitrogen Oxides (NO_x)*. There are no specific requirements relating to NO_x emissions from boilers, incinerators or gas turbine installations.

3.11.2 3.10.2 The total weighted emission value for the ship (WV) is to be calculated as follows:

$$WV_{[\text{ship}]} = \frac{WAEV_{[\text{cert}]} }{WAEV_{[\text{IMO}]} }$$

where

$$WAEV_{[\text{Cert}]} = \frac{\sum_{n=1}^n (\text{NO}_{x[\text{cert}]} \cdot P)}{\sum_{n=1}^n (P)}$$

$$WAEV_{[\text{IMO}]} = \frac{\sum_{n=1}^n (\text{NO}_{x[\text{IMO}]} \cdot P)}{\sum_{n=1}^n (P)}$$

n = the number of individual engines on board the ship

P = the rated power in kW, of each individual installed engine

$NO_x[\text{cert}]$ = the certified NO_x emission value, in g/kWh, for each individual engine

$NO_x[\text{IMO}]$ = the Tier II NO_x emission limit value, in g/kWh of each individual engine, ~~in g/kWh, applicable at the date of construction of the ship, or installation date of the engine, as applicable, as specified in Regulation 13 – Nitrogen Oxides (NOx) of Annex VI to MARPOL.~~

3.11.3 3.10.3 For ships constructed before 1 January 2011, the NO_{-x1} character will be assigned when:

$$\frac{WAEV_{[\text{cert}]}^{x1}}{WAEV_{[\text{IMO}]}^{x1}} \leq 0,80$$

For ships constructed on or after 1 January 2011, the NO_{-x2} character will be assigned when:

$$\frac{WAEV_{[\text{cert}]}^{x2}}{WAEV_{[\text{IMO}]}^{x2}} \leq 0,80$$

3.11.4 3.10.4 For assignment of the NO_{-x3} character, engines, as defined in Pt 7, Ch 11, 2.2 Nitrogen oxides (NOx) 2.2.1, and any associated NO_x emission abatement systems are to be certified as meeting the ~~Tier 3~~ Tier III NO_x emission limits specified in MARPOL Annex VI, Regulation 13 – Nitrogen Oxides (NOx).

Existing paragraphs 3.11.5 to 3.11.9 have been renumbered 3.10.5 to 3.10.9.

Existing sub-Section 3.12 has been renumbered 3.11.

3.13 3.12 Protected oil tanks – P character

Existing paragraph 3.13.1 has been renumbered 3.12.1.

3.13.2 3.12.2 All fuel oil, lubricating oil and hydraulic oil tanks, with a capacity greater than 60 m^3 , ~~Fuel oil, lubricating oil and hydraulic oil tanks~~ are to be located in accordance with the requirements relating to fuel oil tank protection given in MARPOL Annex I, Regulation 12A, paragraphs 6, 7 and 8. Where tanks cannot be located in such locations the vessel is to comply with the accidental fuel oil outflow performance standard, as specified in paragraph 11 of MARPOL, Annex I, Regulation 12A.

Existing paragraphs 3.13.3 to 3.13.5 have been renumbered 3.12.3 to 3.12.5.

Existing sub-Sections 3.14 and 3.15 have been renumbered 3.13 and 3.14.

3.16 3.15 Sulphur oxides – DIST and SOx characters

Existing paragraphs 3.16.1 to 3.16.3 have been renumbered 3.15.1 to 3.15.3.

3.16.4 3.15.4 Alternative arrangements providing an equivalent level of environmental protection will be considered for the assignment of the SOx character. If an Exhaust Gas Cleaning System is fitted, it is to be certified to Resolution MEPC.184(59) – 2009 Guidelines for Exhaust Gas Cleaning Systems – (adopted on 17 July 2009). ~~These systems are to operate at all times regardless of the emission control area in which the vessel operates.~~

Existing sub-Sections 3.17 and 3.18 have been renumbered 3.16 and 3.17.

Part 7, Chapter 13 On-shore Power Supplies

■ Section 3 Electrical Connection

3.3 Connection cables, plugs and socket-outlets

3.3.6 Connection Equipment power cables are to be Type Approved in accordance with LR's *Type Approval System Test Specification Number 3* or, alternatively, surveyed by the Surveyors during manufacture and testing to assess compliance with Pt 7, Ch 13, 3.3 Connection cables, plugs and socket-outlets 3.3.3 and application of an acceptable quality management system, *see also Pt 6, Ch 2, 1.4 Surveys 1.4.6*. Connection equipment cables are to be installed so as to minimise the risk of short-circuit when correctly applied.

■ **Section 6**
Testing, trials and surveys

6.1 General

6.1.3 In addition to *Pt 7, Ch 13, 6.1 General 6.1.2*, the following Connection Equipment, where applicable, is to be surveyed by the Surveyors during manufacture and testing:

- filters;
- converters; and
- slip ring assemblies.

See also *Pt 6, Ch 2, 1.4 Surveys 1.4.6*.

© Lloyd's Register Group Limited 2018
Published by Lloyd's Register Group Limited
Registered office (Reg. no. 08126909)
71 Fenchurch Street, London, EC3M 4BS
United Kingdom

Lloyd's Register and variants of it are trading names of Lloyd's Register Group Limited, its subsidiaries and affiliates. For further details please see
<http://www.lr.org/entities>

Lloyd's Register Group Limited, its subsidiaries and affiliates and their respective officers, employees or agents are, individually and collectively, referred to in this clause as 'Lloyd's Register'. Lloyd's Register assumes no responsibility and shall not be liable to any person for any loss, damage or expense caused by reliance on the information or advice in this document or howsoever provided, unless that person has signed a contract with the relevant Lloyd's Register entity for the provision of this information or advice and in that case any responsibility or liability is exclusively on the terms and conditions set out in that contract.